MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 3, 2018/2019

EME3276 – MECHANICAL DESIGN II

(ME)

1 JUNE 2019
9.00 am - 11.00 am
(2 Hours)

INSTRUCTIONS TO STUDENTS

1.This Question paper consists of 12 pages with 4 Questions only.

2. Attempt **ALL** questions. All questions carry equal marks and the distribution of the marks for each question is given.

3. Please write all your answers in the answer booklet provided.

4. You may use Appendices from page 4 to 12 whenever necessary.

Question 1

a) An angular-contact, inner ring rotating, 02-series ball bearing is having a rating life, X_D equals to 1440. The design load, F_D is radial and equal to 3.8 kN. And the reliability goal is 0.90. Find the catalog rating C_{10}. Use the table, choose a bearing and estimate the existing reliability in service.

[12.5 marks]

b) If another bearing on the shaft is to be a 03-series cylindrical roller bearing with inner ring rotating. The design load, F_D is radial and equal to 10.279 kN And the reliability goal is 0.90. Find the catalog rating C_{10}. Use the table, choose a bearing and estimate the existing reliability in service.

[12.5 marks]

Question 2

A helical valve spring with free end is to be designed for an operating load range of approximately 90 to135 N. The lift, l' of the spring for the load range is 7.5 mm. Assume a spring index of 10, shear strength, $\tau_{max} = 345\ N/mm^2$ and G=79540 N/mm^2. Design the spring for maximum load F_2= 135N, minimum load F_1 =90N; l'=7.5mm, C=10 and find, Maximum deflection, y_2 ; Diameter of wire, d; Mean diameter of the coil, D; Number of turns, N; Solid length, ls; spring rate, k; Total length of wire, l.

[25 marks]

Question 3

a) Two pulleys, one of diameter 150 mm and other of diameter 200 mm are on parallel shafts with centre 600 mm apart.

i. What is the angle of contact between that belt and each pulley?

[9 marks]

ii. What power can be transmitted when larger pulley rotates at 300 rev/min, if maximum tension, T_1 in the belt is to be 1kN and coefficient of friction, μ between belt and pulley is 0.25.

[6 marks]

b) A flat belt is required to transmit 100 kW at a belt speed of 20 m/s between two pulley of diameter 250 mm and 400 mm when the distance between two pulley centre is 1200 mm. The material to be used for the belt has a maximum permissible stress, τ_{max} of 0.8 MPa and is available in thickness to width ratio of 1 to 10. The coefficient of friction, μ between belt and the pulley is 0.3. What is the maximum belt thickness that can be used?

[10 marks]

Continued...

Question 4

A spur gearset with $\varphi = 20°$, and k=1 find the following while avoiding interference.

i) The smallest pinion tooth count that will run with itself

[5 marks]

ii) The smallest pinion tooth count at a ratio, m_G =m= 2.5

[5 marks]

iii) The largest gear tooth counts possible with this pinion

[5 marks]

iv) The smallest pinion that will run with a rack

[5 marks]

v) Find smallest pinion tooth count that will run with itself for a helical gearset with $\varphi_n = 20°$, $\psi = 30°$ and $\varphi_t = 22.8°$.

[5 marks]

Continued...

APPENDIX

EQUATIONS

BEARING

$$x_D = \frac{L}{L_{10}} = \frac{60L_D n_D}{60L_R n_R}$$

$F^a L = constant$

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(ln1/R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

a = 3 for ball bearing and a = 10/3 for roller bearing;
b = 1.483; x_o = 0.02; (θ - x_o) = 4.439 unless otherwise mentioned

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{\frac{1}{b}}}\right]^{\frac{1}{a}}$$

if $R_D \geq 0.9$

$F_e = X_i V F_r + Y_i F_a$

V = 1 for inner ring rotation;
V = 1.2 for outer ring rotation

$$R = exp\left\{-\left(\frac{x_D\left(\frac{a_f F_D}{C_{10}}\right)^a - x_o}{\theta - x_o}\right)^b\right\}$$

$$F_{eq} = \left[\frac{\sum n_i t_i F_{ei}^a}{\sum n_i t_i}\right]^{\frac{1}{a}}$$

SPRING

$$y = \frac{8FD^3N}{d^4G}$$

$$\tau_{max} = K_B \frac{8FD}{\pi d^3}$$

$$k = \frac{F}{y} = \frac{d^4G}{8D^3N}$$

$$K_B = \frac{4C + 2}{4C - 3}$$

$$L_o < \frac{2.63D}{\alpha} \text{ for steel}$$

$$S_{ut} = \frac{A}{d^m}$$

$$C = \frac{D}{d}$$

$L_s = d\,(N + 1)$

$$L_o < \frac{\pi D}{\alpha}\left[\frac{2(E - G)}{2G + E}\right]^{\frac{1}{2}}$$

$L_o = L_s + y$

$p = (L_o - d)/\,N$

$l = \pi DN$

$$y_2 = \frac{y' F_2}{F_2 - F_1}$$

$$F_o = \frac{F_2}{y_2}$$

Continued...

CLUTCHES AND BRAKES

Internal Expanding Rim Clutches and Brakes

$$M_f = \frac{fp_a br}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta(r - a\cos\theta)d\theta = \frac{fp_a br}{\sin\theta_a}\left(-r\cos\theta - \frac{a}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2}$$

$$M_N = \frac{p_a bra}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin^2\theta d\theta = \frac{p_a bra}{\sin\theta_a}\left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

$$T = \frac{fp_a br^2}{\sin\theta_a}\int_{\theta_1}^{\theta_2}\sin\theta d\theta = \frac{fp_a br^2}{\sin\theta_a}(\cos\theta_1 - \cos\theta_2)$$

$$A = \int_{\theta_1}^{\theta_2}\sin\theta\cos\theta d\theta = \left(\frac{1}{2}\sin^2\theta\right)_{\theta_1}^{\theta_2}$$

$$B = \int_{\theta_1}^{\theta_2}\sin^2\theta d\theta = \left(\frac{\theta}{2} - \frac{1}{4}\sin 2\theta\right)_{\theta_1}^{\theta_2}$$

For clockwise rotation:

$$F = \frac{M_N - M_f}{c}$$

$$R_x = \frac{p_a br}{\sin\theta_a}(A - fB) - F_x$$

$$R_y = \frac{p_a br}{\sin\theta_a}(B + fA) - F_y$$

For counterclockwise rotation:

$$F = \frac{M_N + M_f}{c}$$

$$R_x = \frac{p_a br}{\sin\theta_a}(A + fB) - F_x$$

$$R_y = \frac{p_a br}{\sin\theta_a}(B - fA) - F_y$$

Band Type Clutches and Brakes

$$\frac{P_1}{P_2} = e^{f\phi}$$

$$P_1 = p_{\max} br$$

$$T = (P_1 - P_2)\frac{D}{2}$$

Continued.....

Belt

Flat - Open belt

$\theta_d = 180 - 2\ \sin^{-1}\left(\frac{D-d}{2C}\right)$

$\theta_D = 180 + 2\ \sin^{-1}\left(\frac{D-d}{2C}\right)$

$\sin^{-1}\left(\frac{D-d}{2C}\right)$

$P = T_1(1 - e^{-\mu\theta})v$

$v = \omega r$

$T_1 = \tau_{max}\ x\ Area\ of\ belt$

$L = \sqrt{4C^2 - (D-d)^2} + \frac{1}{2}(D\theta_D + d\theta_d)$

Flat - Cross Belt

$\theta = 180 + 2\ \sin^{-1}\left(\frac{D-d}{2C}\right)$

$L = \sqrt{4C^2 - (D+d)^2} + \frac{1}{2}(D+d)\theta$

Gear

$C = (r_1 + r_2) = 0.5\ m\ (Z_1 + Z_2) = 0.5\ m\ (Z_1 + i\ Z_1)$

$d_1 = m\ Z_1\ ;\ d_2 = i\ d_1$

$$Z_1^2 + 2Z_1Z_2 = \frac{4k(Z_2 + k)}{\sin^2\varphi}$$

$$L_a = \sqrt{(r_1 + a)^2 - (r_1^2\cos^2\varphi)} + \sqrt{(r_2 + a)^2 - (r_{12}^2\cos^2\varphi)} - (r_1 + r_2)\sin\varphi$$

$$CR = \frac{L_a}{p\cos\varphi} = \frac{L_a}{\pi m\cos\varphi}$$

Addendum: a = 1m
Dedendum: b = 1.25m
Clearance: c = 0.25 m
Pinion root circle diameter: $d_{r1} = d_1 - 2b$
Pinion addendum diameter: $d_{a1} = d_1 + 2a$
Gear root circle diameter: $d_{r2} = d_2 - 2b$
Gear addendum diameter: $d_{a2} = d_2 + 2a$

Smallest number of teeth for one to one ratio

$$N_P = \frac{2k}{3\sin^2\phi}\left(1 + \sqrt{1 + 3\sin^2\phi}\right)$$

Smallest number of teeth for a gear ratio

$$N_P = \frac{2k}{(1+2m)\sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)\sin^2\phi}\right)$$

Largest number of teeth for pinion ($\varphi = 20^\circ$)

$$N_G = \frac{N_P^2\sin^2\phi - 4k^2}{4k - 2N_P\sin^2\phi}$$

Smallest pinion run with rack

$$N_P = \frac{2(k)}{\sin^2\phi}$$

Smallest number of teeth for helical gearset (with $\varphi_n = 20^\circ$ and $\psi = 30^\circ$, $\varphi_t = 22.8^\circ$)

$$N_P = \frac{2k\cos\psi}{3\sin^2\phi_t}\left(1 + \sqrt{1 + 3\sin^2\phi_t}\right)$$

Continued.....

TABLES
BEARING

Table 1: Dimensions and Basic Load Ratings for Cylindrical Roller Bearings

Bore, mm	02-Series OD, mm	02-Series Width, mm	02-Series Load Rating, kN C_{10}	02-Series Load Rating, kN C_0	03-Series OD, mm	03-Series Width, mm	03-Series Load Rating, kN C_{10}	03-Series Load Rating, kN C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

Table 2: Dimensions and Load Ratings for Single-Row 02-Series Deep Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm d_S	Shoulder Diameter, mm d_H	Load Ratings, kN Deep Groove C_{10}	Load Ratings, kN Deep Groove C_0	Load Ratings, kN Angular Contact C_{10}	Load Ratings, kN Angular Contact C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Continued...

Table 3: Bearing-Life Recommendations for Various Classes of Machinery

Type of Application	Life, kh
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5–2
Machines for short or intermittent operation where service interruption is of minor importance	4–8
Machines for intermittent service where reliable operation is of great importance	8–14
Machines for 8-h service that are not always fully utilized	14–20
Machines for 8-h service that are fully utilized	20–30
Machines for continuous 24-h service	50–60
Machines for continuous 24-h service where reliability is of extreme importance	100–200

Table 4: Load-Application Factors

Type of Application	Load Factor
Precision gearing	1.0–1.1
Commercial gearing	1.1–1.3
Applications with poor bearing seals	1.2
Machinery with no impact	1.0–1.2
Machinery with light impact	1.2–1.5
Machinery with moderate impact	1.5–3.0

Table 5: Formulas for the Dimensional Characteristics of Compression-Springs

Term	Type of Spring Ends			
	Plain	Plain and Ground	Squared or Closed	Squared and Ground
End coils, N_e	0	1	2	2
Total coils, N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length, L_0	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length, L_s	$d(N_t + 1)$	dN_t	$d(N_t + 1)$	dN_t
Pitch, p	$(L_0 - d)/N_a$	$L_0/(N_a + 1)$	$(L_0 - 3d)/N_a$	$(L_0 - 2d)/N_a$

Continued...

Table 6: End-Condition Constants α for Helical Compression Springs

End Condition	Constant α
Spring supported between flat parallel surfaces (fixed ends)	0.5
One end supported by flat surface perpendicular to spring axis (fixed); other end pivoted (hinged)	0.707
Both ends pivoted (hinged)	1
One end clamped; other end free	2

*Ends supported by flat surfaces must be squared and ground.

Table 7: Constants A and m of $S_{ut} = A/d^m$ for Estimating Minimum Tensile Strength of Common Spring Wires

Material	ASTM No.	Exponent m	Diameter, in	A, kpsi · in^m	Diameter, mm	A, MPa · mm^m	Relative Cost of Wire
Music wire*	A228	0.145	0.004–0.256	201	0.10–6.5	2211	2.6
OQ&T wire†	A229	0.187	0.020–0.500	147	0.5–12.7	1855	1.3
Hard-drawn wire‡	A227	0.190	0.028–0.500	140	0.7–12.7	1783	1.0
Chrome-vanadium wire‡	A232	0.168	0.032–0.437	169	0.8–11.1	2005	3.1
Chrome-silicon wire‡	A401	0.108	0.063–0.375	202	1.6–9.5	1974	4.0
302 Stainless wire#	A313	0.146	0.013–0.10	169	0.3–2.5	1867	7.6–11
		0.263	0.10–0.20	128	2.5–5	2065	
		0.478	0.20–0.40	90	5–10	2911	
Phosphor-bronze wire*§	B159	0	0.004–0.022	145	0.1–0.6	1000	8.0
		0.028	0.022–0.075	121	0.6–2	913	
		0.064	0.075–0.30	110	2–7.5	932	

Table 8: Maximum Allowable Torsional Stresses for Helical Spring in Static Applications

Material	Maximum Percent of Tensile Strength Before Set Removed (includes K_W or K_B)	Maximum Percent of Tensile Strength After Set Removed (includes K_s)
Music wire and cold-drawn carbon steel	45	60–70
Hardened and tempered carbon and low-alloy steel	50	65–75
Austenitic stainless steels	35	55–65
Nonferrous alloys	35	55–65

Continued...

Table 9: Mechanical Properties of Some Spring Wires

Material	Elastic Limit, Percent of S_{ut} Tension	Torsion	Diameter d, in	E Mpsi	E GPa	G Mpsi	G GPa
Music wire A228	65–75	45–60	≤0.032	29.5	203.4	12.0	82.7
			0.033–0.063	29.0	200	11.85	81.7
			0.064–0.125	28.5	196.5	11.75	81.0
			≥0.125	28.0	193	11.6	80.0
HD spring A227	60–70	45–55	≤0.032	28.8	198.6	11.7	80.7
			0.033–0.063	28.7	197.9	11.6	80.0
			0.064–0.125	28.6	197.2	11.5	79.3
			≥0.125	28.5	196.5	11.4	78.6
Oil tempered A239	85–90	45–50		28.5	196.5	11.2	77.2
Valve spring A230	85–90	50–60		29.5	203.4	11.2	77.2
Stainless steel							
A313*	65–75	45–55		28	193	10	69.0
17-7PH	75–80	55–60		29.5	208.4	11	75.8
414	65–70	42–55		29	200	11.2	77.2
420	65–75	45–55		29	200	11.2	77.2
431	72–76	50–55		30	206	11.5	79.3

Table 10: Commonly Available Spring Wire Stock Diameters

in	mm	in	mm
0.010	0.25	0.092	
0.012	0.30	0.098	2.50
0.014	0.35	0.105	
0.016	0.40	0.112	2.80
0.018	0.45	0.125	
0.020	0.50	0.135	3.50
0.022	0.55	0.148	
0.024	0.60	0.162	4.00
0.026	0.65	0.177	4.50
0.028	0.70	0.192	5.00
0.030	0.80	0.207	5.50
0.035	0.90	0.225	6.00
0.038	1.00	0.250	6.50
0.042	1.10	0.281	7.00
0.045		0.312	8.00
0.048	1.20	0.343	9.00
0.051		0.362	
0.055	1.40	0.375	
0.059		0.406	10.0
0.063	1.60	0.437	11.0
0.067		0.469	12.0
0.072	1.80	0.500	13.0
0.076		0.531	14.0
0.081	2.00	0.562	15.0
0.085	2.20	0.625	16.0

[1]Custom wire sizes also available at extra cost.

Continued...

Table 11 Safe Static Stress for use in the Lewis Equation

Material	[σ] MPa	BHN
Gray cast iron		
ASTM 25	122	174
ASTM 35	183	212
ASTM 50	228	223
Cast steel (low carbon)		
0.20% C not heat treated	304	180
0.20% C WQT	380	250
Forged carbon steel		
SAE 1020 case hardened and WQT	274	156
SAE 1030 not heat treated	304	180
SAE 1035 not heat treated	350	190
SAE 1040 not heat treated	380	202
SAE 1045 not heat treated	456	205
SAE 1045 hardened by WQT	487	215
SAE 1050 hardened by OQT	532	223
Alloy Steels		
SAE 2320 case hardened and WQT	761	225
SAE 2345 hardened by OQT	761	475
SAE 3115 case hardened and OQT	563	212
SAE 3145 hardened by OQT	806	475
SAE 3245 hardened by OQT	989	475
SAE 4340 hardened by OQT	989	475
SAE 4640 hardened by OQT	837	475
SAE 6145 hardened by OQT	1019	475

Table 1 VALUES OF THE LEWIS FORM FACTOR Y

Number of teeth	Ø = 20° a = 0.8m* b = m	Ø = 20° a = m b = 1.25m	Ø = 25° a = m b = 1.25m
12	0.335 12	0.229 60	0.276 77
13	0.348 27	0.243 17	0.292 81
14	0.359 85	0.255 30	0.307 17
15	0.370 13	0.266 22	0.320 09
16	0.379 31	0.276 10	0.331 78
17	0.387 57	0.285 08	0.342 40
18	0.395 02	0.293 27	0.352 10
19	0.401 79	0.300 78	0.360 99
20	0.407 97	0.307 69	0.369 16
21	0.413 63	0.314 06	0.376 71
22	0.418 83	0.319 97	0.383 70

Table 1 VALUES OF THE LEWIS FORM FACTOR Y

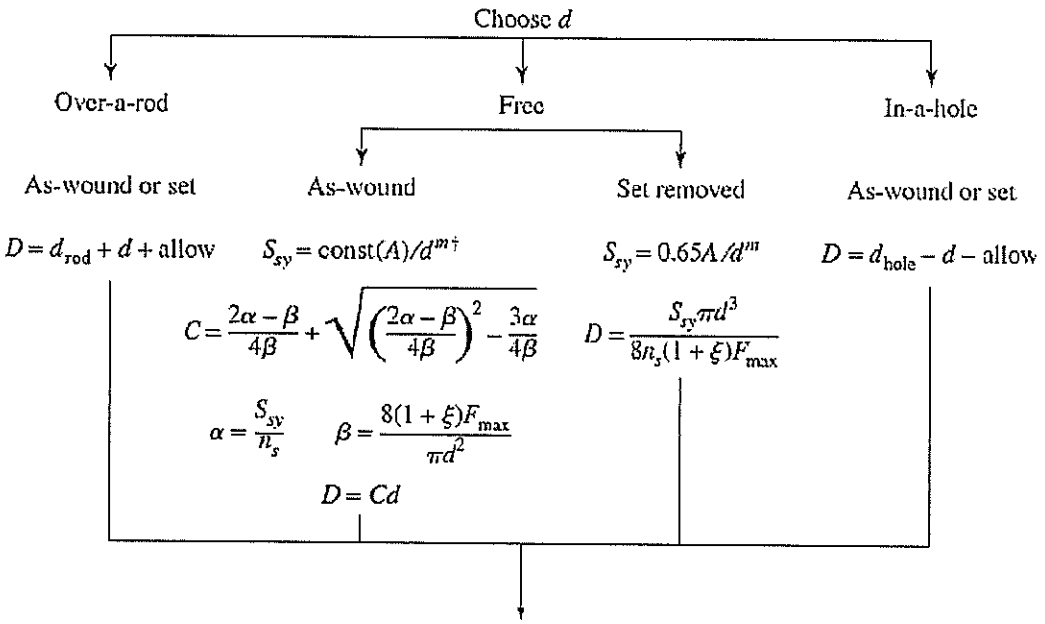
Number of teeth	Ø = 20° a = 0.8m* b = m	Ø = 20° a = m b = 1.25m	Ø = 25° a = m b = 1.25m
24	0.428 06	0.330 56	0.396 24
26	0.436 01	0.339 79	0.407 17
28	0.442 94	0.347 90	0.416 78
30	0.449 02	0.355 10	0.425 30
34	0.459 20	0.367 31	0.439 76
38	0.467 40	0.377 27	0.451 56
45	0.478 46	0.390 93	0.467 74
50	0.484 58	0.398 60	0.476 81
60	0.493 91	0.410 47	0.490 86
75	0.503 45	0.422 83	0.505 46
100	0.513 21	0.435 74	0.520 71

Continued...

FIGURES

SPRING

Figure 1: Design Flowchart for Static Loading



$$C = D/d$$

$$K_B = (4C + 2)/(4C - 3)$$

$$\tau_s = K_B 8(1 + \xi)F_{max}D/(\pi d^3)$$

$$n_s = S_{sy}/\tau_s$$

$$\text{OD} = D + d$$

$$\text{ID} = D - d$$

$$N_a = Gd^4 y_{max}/(8D^3 F_{max})$$

$$N_t$$

$$L_s$$

$$L_0$$

$$(L_0)_{cr} = 2.63D/\alpha$$

$$\text{fom} = -(\text{rel. cost})\gamma\pi^2 d^2 N_t D/4$$

End of paper